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Toward to the control system of mechanical draft cooling tower of film type

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Abstract

For changing atmospheric conditions the mathematical model of a control system of a mechanical draft cooling tower has been developed. The model includes a heat and mass transfer processes between water films and turbulent damp air flow at quasi-state approximation. Various regimes of cooling tower performance are compared and the optimization method is proposed.

1. Introduction

Mechanical draft cooling towers are used at vast majority of industrial mills for cooling the circulating water, which have been used at some technological units. In such cooling tower (Fig. 1) a hot water to be cooled is sprayed by nozzles and flowing down in the pack. As the water drops downward onto the pack fan pull air across the flowing water to remove the heat.

2. Mathematical model of evaporative cooling

As background for simulation of the control system of mechanical draft cooling tower we use our mathematical model of evaporative cooling of water films [1]. It should be mentioned that at [1] we used heat and mass transfer coefficients obtained for laminar air flow along interfacial surface [2]. Due to higher air velocities in mechanical draft cooling towers we have to use heat and mass transfer coefficients, which are valid for turbulent flow. For this purpose we use available experimental data from [2] and well-known similarity between heat and mass transfer processes.

For quasi-state approximation of atmospheric conditions our mathematical model of evaporative cooling of water films [1] represents itself the boundary-value problem for the system of ordinary differential equations. These equations are obtained from equations of mass and energy conservation by averaging in cross direction of the flow. We have the following equations.

The equation describing the change in the film thickness $h(z)$ due to evaporation

$$\frac{\mathrm{d}h(z)}{\mathrm{d}z} = -\frac{\gamma_{\mathrm{f}}(Re)(\rho_{\mathrm{s}}(T_{\mathrm{f}}(z)) - \rho_{\mathrm{v}}(z))}{\rho_{\mathrm{w}} v_{\mathrm{f}}}, \quad (1)$$

the equation determining the change in the cross-section averaged water film temperature $T_{\mathrm{f}}(z)$ due to the contact with the cold air and due to evaporation

$$\frac{\mathrm{d}T_{\mathrm{f}}(z)}{\mathrm{d}z} = \frac{\alpha_{\mathrm{f}}(Re)(T_{\mathrm{a}}(z) - T_{\mathrm{f}}(z)) - r\gamma_{\mathrm{f}}(Re)(\rho_{\mathrm{s}}(T_{\mathrm{f}}(z)) - \rho_{\mathrm{v}}(z))}{c_{\mathrm{w}}\rho_{\mathrm{w}} h(z) v_{\mathrm{f}}}, \quad (2)$$

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Nomenclature

c	specific heat ($\mathrm{J\,kg^{-1}\,K^{-1}}$)
d	distance between two nearest plates of the pack (m)
d_1	distance between two films on nearest plates of the pack (m)
D	diffusion coefficient for water vapor ($\mathrm{m^2/s}$)
F	water flow rate ($\mathrm{m^3/s}$)
g	free fall acceleration ($\mathrm{m/s^2}$)
H	height (m)
h	film thickness (m)
Q_a	specific air mass flow rate (kg/m s)
Q_w	specific water mass flow rate (kg/m s)
r	latent heat of vaporization ($\mathrm{kJ\,kg^{-1}}$)
T	temperature (°C)
v	velocity (m/s)
z	coordinate in vertical direction
Re	Reynolds number

Greek symbols

α	heat transfer coefficient, ($\mathrm{W\,m^{-2}\,K^{-1}}$)
γ	mass transfer coefficient ($\mathrm{m\,s^{-1}}$)
η	thermal efficiency of a cooling tower, dimensionless
λ_a	thermal conductivity of air, (W/m °C)
μ_a	dynamic viscosity of air ($\mathrm{kg\,m^{-1}\,s^{-1}}$)
ν	kinematic viscosity ($\mathrm{m^2/s}$)
ρ	density ($\mathrm{kg/m^3}$)
τ	time of equalizing (s)
ψ	relative humidity of air, dimensionless
Σ	residual, dimensionless
Ω	pond volume ($\mathrm{m^3}$)

Subscripts

0	initial
a	air
f	film
lim	limiting
out	final
s	saturated
v	vapor
w	water

the equation for calculating the change in the cross-section averaged temperature of the moist air $T_a(z)$

$$\frac{\mathrm{d}T_a(z)}{\mathrm{d}z} = -\frac{2\alpha_f(Re)(T_f(z) - T_a(z))}{v_a d_1 \rho_a c_a}, \tag{3}$$

the equation describing the change in the density of the water vapor $\rho(z)$ in air, rising between plates of the pack

$$\frac{\mathrm{d}\rho_v(z)}{\mathrm{d}z} = -\frac{2\gamma_f(Re)(\rho_s(T_f(z)) - \rho_v(z))}{v_a d_1}. \tag{4}$$

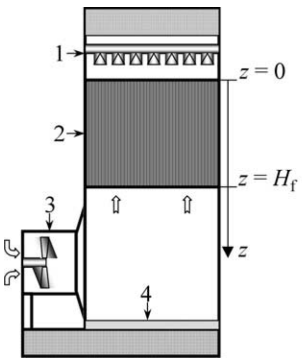

Fig. 1. Scheme of mechanical draft cooling tower: (1) water distribution system with spraying nozzles, (2) film pack, (3) fan and (4) water-collecting pond.

We note that $d_1 = d - 2h_f$ where h_f is the thickness of water film. Therefore d_1 slightly depends on a hydraulic load of a cooling tower.

The system of differential Eqs. (1)–(4) has to be integrated with the following boundary conditions, which describes the directions and inlets of all flow.

At $z = 0$ we have inlet for water and the condition of constant temperature:

$$h|_{z=0} = h_0, \tag{5}$$

$$T_f|_{z=0} = T_{f0}. \tag{6}$$

At $z = H_f$ we have inlet for air and the conditions of temperature and density of water vapor:

$$T_a|_{z=H_f} = T_{a0}, \tag{7}$$

$$\rho_v|_{z=H_f} = \rho_{sv}(T_{a0}) \cdot \psi = \rho_{v0}. \tag{8}$$

In Eqs. (2) and (3), $\alpha(Re)$ is the heat-transfer coefficient determined for turbulent regime of air flow over water interfacial surface [2]:

$$\alpha_f = \frac{0.025 Re^{0.8} \lambda_a}{H_f - z}. \tag{9}$$

For the problem under investigation the Reynolds number is determined [2,3] as following

$$Re = \frac{(H_f - z)\rho_a(v_a + v_f)}{\mu_a}. \tag{10}$$

Using the similarity between the heat transfer and mass transfer processes, we determine the mass-transfer coefficient between the turbulent air flow and a thin film of water γ_f in Eqs. (1) and (4) as follows:

$$\gamma_f = \frac{0.025 Re^{0.8} D}{H_f - z}. \tag{11}$$

In our calculations, we take into account the temperature dependence of the diffusion coefficient of water vapor D and water viscosity [4].

The velocity of the descending water film, averaged over the cross section, was determined in a laminar approximation [3,5] is:

$$v_f = \left(\frac{g}{3\nu_w}\right)^{\frac{1}{3}} \left(\frac{Q_w}{\rho_w}\right)^{\frac{2}{3}}. \tag{12}$$

The boundary-value problem (1)–(8) for the system of nonlinear ordinary differential Eqs. (1)–(4) is solved by a "shooting" method [6]. The procedure of obtaining of numerical solution of the system of the differential equations is based on the Runge–Kutta method of the fourth order. The accuracy of calculations is controlled by means of the residual criterion Σ_f:

$$\Sigma_f = \sqrt{\left(\frac{T_a(H_f) - T_{a0}}{T_{a0}}\right)^2 + \left(\frac{\rho_v(H_f) - \rho_{v0}}{\rho_{v0}}\right)^2}. \tag{13}$$

The solution was terminated as soon as the condition $\Sigma_f < 10^{-4}$ was satisfied. A further increase in the accuracy of calculations did not influence on calculated water temperature in the pool of cooling tower.

3. Some results of simulation

In our publications [1,7] we describe the efficiency of evaporative cooling of water by means of the thermal efficiency of a cooling tower:

$$\eta = \frac{T_{w0} - T_{w_{out}}}{T_{w0} - T_{lim}}, \tag{14}$$

where T_{w0} is the temperature of hot water entering the cooling tower, $T_{w\ out}$ is the temperature of water leaving the cooling tower, T_{lim} is the limiting temperature for evaporative cooling of water, which is equal to the wet-bulb temperature. T_{lim} can be also obtained from the equation

$$\rho_s(T_{lim}) = \psi \cdot \rho_s(T_a), \tag{15}$$

where ρ_s is the density of saturated vapor, ψ is the relative humidity of air, T_a is the temperature of the surrounding air [8].

It was mentioned [1] that the efficiency of cooling of the water film depends substantially on the ratio of water and air mass flow rates. Our numerical experiments show that the thermal efficiency of the pack is nonlinear, monotonically decreasing function of the relation of the specific mass flow rates of the water and the air, Q_w/Q_a.

For mechanical draft cooling tower with heat and mass transfer coefficient, which include turbulent regime of flow, the dependence of the thermal efficiency η on the ratio between the mass flow rates of water and air, Q_w/Q_a, is shown in Fig. 2. This result was obtained for $T_{w0} = 40\,°C$, $q_w = 0.085\,kg/(m\,s)$, $T_{a0} = 20\,°C$ $(15\,°C)$, $\psi = 0.65$, $H = 2\,m$, under changing air flow rate. As Q_w/Q_a ratio increases, the thermal efficiency of a cooling tower decreases. This is the typical property of all cooling towers. For a mechanical draft cooling tower of spray class, the dependence of η versus Q_w/Q_a was obtained in [7].

For given relative humidity the curve 1 is higher then curve 2 only because of the dependence of T_{lim} on T_a (see Eq. (15)). Our calculations show that the flow rate of evaporated water vapor is about 1% of water flow rate Q_w.

3.1. Control system

Optimization of the cooling tower performance is one of the most important problems in the theory and engineering practice of evaporative cooling of water because it permits significantly decreases energy consumption. Some theoretical aspects of this problem for various technological processes are discussed in [9] and for spray mechanical draft cooling tower are in [7].

Let us consider optimization of the performance of a mechanical draft cooling tower of film class in the following formulation: for the given constant water temperature drop it is required to determine the air flow rate through the cooling tower. For simplicity the initial temperature of circulating water and its flow rate are considered constant also, but the temperature and humidity of air are variable quantities. Constrains of a constant initial water temperature and a constant flow rate can be easily eliminated from our mathematical model. For determining the minimal air flow rate, the

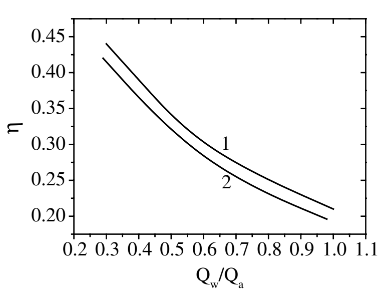

Fig. 2. Thermal efficiency η vs. Q_w/Q_a: curve 1 is for entering air temperature $T_a = 20\,°C$, curve 2 is for entering air temperature $T_a = 15\,°C$.

described above mathematical model was solved starting from small value of air velocity. For given atmospheric conditions and hydraulic load we seek the minimal air velocity, which leads to given drop of water temperature.

As illustration we consider one specific problem, which has the industrial application. Let the initial water temperature is equal to 40 °C, final temperature of water has to be exactly 30 °C. Deeper cooling is not necessary and economically is unjustified. The water flow rate per unit of length is equal to 0.085 kg/(m s). These values correspond to many mechanical draft cooling towers, which are widely used at industry. The results of calculation, obtained by means of our approach, are presented in Fig. 3. As it seen from Fig. 3, with increase in the humidity of the atmospheric air it is necessary to increase the air flow rate through cooling tower to reach the same given water temperature drop. For optimization of the energy consumption it is possible to decrease the fan power as the air temperature decreases. To remind the fan power is directly proportional to cube the of air velocity. It is worth to note that the decrease of air flow rate permits to save significant amount circulating technical water due to evaporation.

We use quasy-state approximation at our mathematical model, therefore it is useful to know the time of equalizing of temperature in the pond of cooling tower. If Ω is the pond volume, then time of equalization of temperature τ we will be consider equal to 3τ

$$\tau = 3\Omega/F_{\mathrm{w}}. \tag{16}$$

For relatively small cooling towers typical value of τ is about 2–5 min. For vast majority processes in atmosphere duration of change in any atmospheric parameter is greater than τ, we with full confidence can consider the cooling tower performance as quasy-stationary and use the steady-state solutions of our mathematical model.

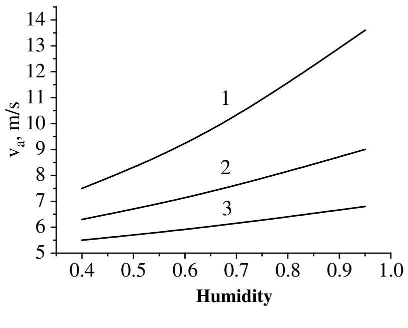

Fig. 3. Air velocity v_{a} vs. relative humidity ψ for given water temperature drop 10 °C. Curve 1 is for entering air temperature T_{a} = 25 °C, curve 2–20 °C, curve 3–15 °C.

Table 1
Optimal air velocities for two cooling towers

Humidity	70	80	90	100
Air velocity	4.4	4.8	5.5	6.5

3.2. Different options

If a fan power is not big enough in order to reach the given temperature drop, we have to choose another methods of organization of cooling process. Here we will consider only two additional approaches.

The first one is to decrease water flow rate per cooling tower and switch on additional cooling tower. For example, let the maximum air velocity in the cooling tower be 10 m/s. Our model helps to reckon the air velocity for twice smaller water flow rate (for simplicity we assume the water flow was symmetrically divided between cooling towers). For air temperature 25 °C the values of calculated air velocity are shown in table.

As it is seen from Table 1 in order to reach the same water temperature drop 10 °C under increasing humidity it is needed to divide water flow between two cooling towers and change air velocity from 6 m/s to the values shown in table. If air velocity will remain 6 m/s then the water temperature drop will fall from required 10 °C to 7.7 °C. If air humidity is constant and equals to 65% then, using such technology water will be cooled to required temperature drop at air velocity 4 m/s.

There is another option. If the circulating water is not divided between two cooling towers and all water flow is cooled in first tower and then cooled in second one. Then for air velocity 4 m/s the first cooling tower will give water temperature drop 5.9 °C and the second one −3.8 °C. And the total temperature drop in both towers will be 9.7 °C.

So in our case the way of dividing of flow rate is slightly more effective due to the influence of initial water temperature on its cooling. It should be noticed that this method is useful for cooling towers where distribution of water by spraying nozzles is not dependent on water flow rate.

The third option is to add fresh relatively cold water in the pond of cooling tower and to use cooled mixture. Our model [1] helps to calculate this scenario, using additional information about the cost of water and cooling tower performance. In particular, how to minimize the flow rate of additional water with given temperature.

4. Conclusions

For the film type counter-flow mechanical draft cooling tower mathematical model of the control system has been developed. It based on our approach to simulation of cooling tower performance [1,7]. The problem of cooling tower performance optimization under varying atmospheric conditions was solved. Parallel and step-by step cooling methods in two towers were considered. It was shown that that parallel cooling is slightly more effective. But in case when the performance of spraying nozzle depends on water flow rate step-by step cooling method should be organized.

Use of the control system permits to provide the drop of water temperature under changing atmospheric conditions. When deeper cooling is not necessary, the control system decreases energy consumption and the amount of evaporated water.

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